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Description

REFRIGERATION SYSTEM

Technical Field

[0001] This invention relates to refrigeration systems that operate in a refrigeration cycle having a high-side refrigerant pressure set equal to or above the critical pressure of the refrigerant.

Background Art

[0002] Refrigeration systems have been conventionally known that operate in a refrigeration cycle by circulating refrigerant through a refrigerant circuit that is a closed circuit and are widely used, for example, as air conditioners. As disclosed, for example, in Patent Document 1, there is known a refrigeration system of such kind in which the high-side refrigerant pressure of a refrigeration cycle is set above the critical pressure of the refrigerant. The refrigeration system includes an expander as an expansion mechanism for the refrigerant. Further, in the refrigeration system, the expander is connected to a compressor by a shaft and energy obtained from the expander is used to drive the compressor, thereby improving the COP (coefficient of performance).

Patent Document 1: Unexampled Japanese Patent Publication No. 2001-107881

Disclosure of the Invention

Problems to be Solved by the Invention

[0003] In designing the above refrigeration system with an expander, the specification of the expander must be defined. In particular, in the case of using as the expander a displacement fluid machine of a type in which a fluid chamber forming a closed space is variable in volume, it is necessary to specify the volume of the fluid chamber just after the

closing off of fluid communication from its inlet channel (i.e., at the starting point of fluid expansion in the fluid chamber) and the volume of the fluid chamber just before the provision of fluid communication with its outlet channel (i.e., at the end point of fluid expansion in the fluid chamber). In order to specify these values, it is necessary to previously set the refrigerant conditions at both the entrance and exit of the expander. In the case where the expander and the compressor are connected by a shaft or the like so that the rotational speed ratio between them is fixed, it is also necessary to consider the volume of the compressor.

[0004] Then, when the capacity desired for the refrigeration system is specified, the design value of the volume of the compressor can be set accordingly. Further, when the application of the refrigeration system is specified, an object to be given heat by the refrigerant (a heat-given object) and an object to be taken heat by the refrigerant (a heat-taken object) can be assumed. For example, where the refrigeration system forms an air conditioner and when it is in cooling operation, the heat-given object is outdoor air and the heat-taken object is room air.

[0005] When the temperature condition of the heat-given object is fixed, as shown in the Mollier diagram (pressure-enthalpy diagram) of Figure **11**, the evaporation temperature T_1 of the refrigerant can be set accordingly and, in turn, the design value of the low-side pressure P_L of a refrigeration cycle can be determined. On the other hand, when the high-side pressure of the refrigeration cycle reaches or exceeds the critical pressure of the refrigerant, the high-pressure refrigerant releases heat but does not condense. Therefore, even when the temperature condition of the heat-given object is fixed, only the refrigerant temperature T_2 after heat release can be set accordingly. In other words, it is determined only that the refrigerant condition after heat release lies on the isothermal curve for the temperature T_2. Hence, the design value of the high-side pressure of the refrigeration cycle cannot be uniquely determined.

[0006] As described above, in the refrigeration system with an expander in which the

high-side refrigerant pressure of a refrigeration cycle reaches or exceeds the critical pressure of the refrigerant, the design value of the high-side pressure of the refrigeration cycle cannot be set simply by specifying the application of the refrigeration system. Further, if the high-side pressure of the refrigeration cycle is not fixed, then the refrigerant condition at the entrance of the expander (the condition expressed by point X in Figure **11**) can not be set and, in turn, the refrigerant condition at the exit of the expander (the condition expressed by point Y in Figure **11**) can not be set. Therefore, the known refrigeration system has a problem that the specification of the expander does not necessarily meet the use conditions of the refrigeration system and efficiency improvement due to the expander cannot sufficiently be provided.

[0007] The present invention has been made in view of the foregoing points and, therefore, its object is to provide a refrigeration system that includes an expander meeting the operating conditions of the refrigeration system and provides high efficiency.

Means for Solving the Problems

[0008] A first solution of the invention is directed to a refrigeration system including a refrigerant circuit **(15)** in which a compressor **(50)**, a gas cooler **(16)**, an expander **(60)** and an evaporator **(17)** are connected, the refrigeration system operating in a refrigeration cycle having a high-side refrigerant pressure equal to or above the critical pressure of the refrigerant by circulating the refrigerant through the refrigerant circuit **(15)**. Further, the compressor **(50)** and the expander **(60)** are each composed of a displacement fluid machine whose fluid chamber is variable in volume and are connected one to the other with the rotational speed ratio of the one to the other fixed, and the volume v_2 of the fluid chamber in the expander **(60)** just after the closing off of fluid communication from an inlet channel thereof is set to $v_2=\rho_1 v_1 r/\rho_2$ and the volume v_3 of the fluid chamber in the expander **(60)** just before the provision of fluid communication with an outlet channel thereof is set to $v_3=\rho_2 v_2/\rho_3$, where: the low-side refrigerant pressure of the refrigeration cycle and the refrigerant temperature at the exit of the gas cooler **(16)** under reference operating

conditions serving as a design standard are employed as a reference low pressure and a reference refrigerant temperature, respectively; the high-side refrigerant pressure of the refrigeration cycle at which the coefficient of performance of the refrigeration cycle reaches a maximum value under the reference operating conditions is employed as a reference high pressure; ρ_1 is the density of saturated gas refrigerant at the reference low pressure; ρ_2 is the density of refrigerant at the reference high pressure and the reference refrigerant temperature; ρ_3 is the density of refrigerant adiabatically expanded from a condition of the reference high pressure and the reference refrigerant temperature into a condition of the reference low pressure; v_1 is the volume of the fluid chamber in the compressor (**50**) just after the closing off of fluid communication from a suction channel thereof; and r is the rotational speed ratio of the compressor (**50**) to the expander (**60**).

[0009] A second solution of the invention is directed to the first solution and characterized in that the refrigerant circuit (**15**) is provided with a receiver (**18**) between the exit side of the evaporator (**17**) and the suction side of the compressor (**50**).

[0010] A third solution of the invention is directed to the first solution and characterized in that the refrigerant circuit (**15**) is provided with an internal heat exchanger (**20**) for providing heat exchange between refrigerant flowing from the gas cooler (**16**) towards the expander (**60**) and refrigerant flowing from the evaporator (**17**) towards the compressor (**50**).

[0011] – Behaviors –

In the first solution, the refrigeration system (**10**) operates in a refrigeration cycle by circulating refrigerant through the refrigerant circuit (**15**). The refrigerant circulating through the refrigerant circuit (**15**) is compressed into a supercritical state by the compressor (**50**), then releases heat at the gas cooler (**16**), expands in the expander (**60**), takes heat at the evaporator (**17**) to evaporate, is sucked into the compressor (**50**) and compressed therein again. The expander (**60**) is connected to the compressor (**50**) so that energy recovered from the refrigerant by the expander (**60**) is used to drive the compressor

(50). In the refrigeration system (10), the rotational speed ratio of the expander (60) to the compressor (50) is fixed. For example, the rotational speed ratio is "1" when the compressor (50) and the expander (60) are directly connected by a single shaft. When they are connected via a speed reducer, the rotational speed ratio is equal to the reduction gear ratio of the speed reducer.

[0012] In designing the refrigeration system (10), the volume v_l of the fluid chamber in the compressor (50) just after the closing off of fluid communication from its suction channel, i.e., the suction volume of the compressor (50), can be set according to the desired value of its capacity corresponding to the refrigerating capacity. Further, when the application of the refrigeration system (10) is specified, the low-side pressure of the refrigeration cycle, i.e., the reference low pressure, and the refrigerant temperature at the exit of the gas cooler (16), i.e., the reference refrigerant temperature, can be determined based on operating conditions expected from the application. However, the high-side pressure of the refrigeration cycle cannot be determined from only the reference low pressure and the reference refrigerant temperature. When the low-side pressure of the refrigeration cycle and the refrigerant temperature at the exit of the gas cooler (16) are set, the high-side pressure of the refrigeration cycle at which the coefficient of performance (COP) reaches a maximum value under the operating conditions can be determined through an operating test or a simulation under the operating conditions.

[0013] To achieve this, in this solution, under the reference operating conditions in which the low-side pressure of the refrigeration cycle and the refrigerant temperature at the exit of the gas cooler (16) are a reference low pressure and a reference refrigerant temperature, respectively, the high-side pressure of the refrigeration cycle at which the coefficient of performance thereof reaches a maximum value is employed as a reference high pressure. Further, the specification of the expander (60) in the refrigeration system (10) is defined based on the reference low pressure, the reference refrigerant temperature, the reference high pressure and the material properties of refrigerant charged in the

refrigerant circuit **(15)**. Specifically, the volume v_2 of the fluid chamber in the expander **(60)** just after the closing off of fluid communication from its inlet channel (i.e., just before the start of refrigerant expansion in the fluid chamber) and the volume v_3 of the fluid chamber in the expander **(60)** just before the provision of fluid communication with its outlet channel (i.e., just after the end of refrigerant expansion in the fluid chamber) are set based on the above factors including the reference high pressure.

[0014] In the second solution, the refrigerant circuit **(15)** is provided with a receiver **(18)**. Within the receiver **(18)**, part of refrigerant charged in the refrigerant circuit **(15)** is stored in the form of liquid refrigerant. When the amount of liquid refrigerant in the receiver **(18)** increases or decreases, the amount of refrigerant circulating through the refrigerant circuit **(15)** varies accordingly. The receiver **(18)** is disposed between the exit side of the evaporator **(17)** and the suction side of the compressor **(50)**. In the refrigerant circuit **(15)**, refrigerant having flowed out of the evaporator **(17)** is introduced into the receiver **(18)** and refrigerant in the receiver **(18)** is sucked into the compressor **(50)**. Since liquid refrigerant exists in the receiver **(18)**, refrigerant in the receiver **(18)** to be sucked into the compressor **(50)** is in a saturated state.

[0015] In the third solution, the refrigerant circuit **(15)** is provided with an internal heat exchanger **(20)**. In the internal heat exchanger **(20)**, refrigerant flowing from the gas cooler **(16)** towards the expander **(60)** is cooled by heat exchange with refrigerant flowing from the evaporator **(17)** towards the compressor **(50)**. The refrigerant flowing towards the expander **(60)** decreases its enthalpy by being cooled in the internal heat exchanger **(20)**. The refrigerant sent from the expander **(60)** to the evaporator **(17)** also decreases its enthalpy accordingly.

Effects of the Invention

[0016] In the present invention, attention is focused on a characteristic of the refrigeration system **(10)** that under conditions in which the low-side pressure of a refrigeration cycle (the reference low pressure) and the refrigerant temperature at the exit

of the gas cooler **(16)** (the reference refrigerant temperature) are fixed, the high-side pressure of the refrigeration cycle (the reference high pressure) at which the coefficient of performance (COP) reaches its maximum value is uniquely determined, and the expander **(60)** specified using the above characteristic is provided in the refrigeration system **(10)**. Therefore, according to the present invention, the expander **(60)** having a specification meeting the use conditions of the refrigeration system **(10)** can be used and energy can be reliably recovered from refrigerant by the expander **(60)** to enhance the COP of the refrigeration system **(10)**.

[0017] In the second solution, the refrigeration system is configured to introduce refrigerant having flowed out of the evaporator **(17)** into the receiver **(18)** and allow the compressor **(50)** to suck refrigerant from the receiver **(18)**. Therefore, even under an operating condition in which refrigerant flowing out of the evaporator **(17)** is in a superheated state, the compressor **(50)** sucks saturated refrigerant in the receiver **(18)**. Hence, according to this solution, the condition of refrigerant to be sucked into the compressor **(50)** can be matched with the condition expected in designing the refrigeration system **(10)**, which enables stabilization of the refrigeration cycle in the refrigerant system **(10)**.

[0018] In the third solution, the refrigerant circuit **(15)** is provided with the internal heat exchanger **(20)** to cool refrigerant flowing into the expander **(60)** and thereby decrease the enthalpy of refrigerant sent from the expander **(60)** to the evaporator **(17)**. Therefore, the refrigerant enthalpy difference between the entrance and exit of the evaporator **(17)** can be increased, thereby improving the cooling capacity when the object is cooled in the evaporator **(17)**.

Brief Description of Drawings

[0019] [Figure 1] Figure **1** is a schematic block diagram showing a refrigerant circuit of an air conditioner of Embodiment 1.

[Figure 2] Figure **2** is a schematic cross-sectional view of a compression/expansion unit in Embodiment 1.

[Figure 3] Figure **3** is an enlarged view of an essential part of an expander in Embodiment 1.

[Figure 4] Figure **4** illustrates cross-sectional views showing the states of rotary mechanisms of the expander in Embodiment 1 at every 90° of shaft rotation.

[Figure 5] Figure **5** is a Mollier diagram (pressure-enthalpy diagram) showing a refrigeration cycle in the refrigerant circuit.

[Figure 6] Figure **6** is a graph showing the relation between high-side pressure and COP when the low-side pressure and the refrigerant temperature at the exit of the gas cooler in a supercritical cycle are fixed.

[Figure 7] Figure **7** is a schematic block diagram showing a refrigerant circuit of an air conditioner of Embodiment 2.

[Figure 8] Figure **8** is a schematic block diagram showing a refrigerant circuit of an air conditioner of Embodiment 3.

[Figure 9] Figure **9** is a schematic block diagram showing a refrigerant circuit of an air conditioner in a modification of Embodiment 3.

[Figure 10] Figure **10** illustrates schematic cross-sectional views showing the structure and behaviors of an expander in Embodiment 4.

[Figure 11] Figure **11** is a Mollier diagram (pressure-enthalpy diagram) showing a characteristic of a supercritical cycle.

Explanation of Reference Numerals

[0020] **15** refrigerant circuit

16 gas cooler

17 evaporator

18 receiver

20 internal heat exchanger

50	compressor
53	compression chamber
60	expander
72	first fluid chamber
73	first high-pressure chamber
74	first low-pressure chamber
82	second fluid chamber
83	second high-pressure chamber
84	second low-pressure chamber
95	first chamber
96	second chamber

Best Mode for Carrying Out the Invention

[0021] Embodiments of the present invention will be described below in detail with reference to the drawings.

[0022] <<Embodiment 1 of the Invention>>

A description is given to Embodiment 1 of the present invention. An air conditioner **(10)** of the present embodiment is composed of a refrigeration system according to the present invention.

[0023] As shown in Figure **1**, the air conditioner **(10)** includes a refrigerant circuit **(15)**. The refrigerant circuit **(15)** is charged with carbon dioxide (CO_2) as a refrigerant. Further, the refrigerant circuit **(15)** is provided with a gas cooler **(16)**, an evaporator **(17)** and a compression/expansion unit **(30)**. The gas cooler **(16)** is connected on the entrance side to a discharge pipe **(36)** for the compression/expansion unit **(30)** and connected on the exit side to an inlet port **(34)** of the compression/expansion unit **(30)**. The evaporator **(17)** is connected on the entrance side to an outlet port **(35)** of the compression/expansion unit

(30) and connected on the exit side to a suction port (32) of the compression/expansion unit (30). The gas cooler (16) and the evaporator (17) allow the refrigerant in the refrigerant circuit (15) to exchange heat with air.

[0024] As shown in Figure 2, the compression/expansion unit (30) includes a casing (31) that is a vertically long, cylindrical, closed container. Inside the casing (31), a compressor (50), an electric motor (45) and an expander (60) are arranged in bottom to top order.

[0025] The discharge pipe (36) is attached to the casing (31). The discharge pipe (36) is located between the electric motor (45) and the expander (60) and communicated with the inner space of the casing (31).

[0026] The electric motor (45) is disposed inside a longitudinally middle portion of the casing (31). The electric motor (45) is formed of a stator (46) and a rotor (47). The stator (46) is fixed to the casing (31). The rotor (47) is placed inside the stator (46). Further, a main spindle (44) of a shaft (40) concentrically passes through the rotor (47).

[0027] The shaft (40) constitutes a rotation shaft. The shaft (40) is formed at its lower end with two lower eccentric parts (58, 59) and formed at its upper end with two large-diameter eccentric parts (41, 42).

[0028] The two lower eccentric parts (58, 59) are formed to have a larger diameter than the main spindle (44), the lower of them constitutes a first lower eccentric part (58) and the upper constitutes a second lower eccentric part (59). The first lower eccentric part (58) and the second lower eccentric part (59) have opposite directions of eccentricity with respect to the axis of the main spindle (44).

[0029] The two large-diameter eccentric parts (41, 42) are formed to have a larger diameter than the main spindle (44), the lower of them constitutes a first large-diameter eccentric part (41) and the upper constitutes a second large-diameter eccentric part (42). The first large-diameter eccentric part (41) and the second large-diameter eccentric part (42) have the same direction of eccentricity. The second large-diameter eccentric part (42) has a larger outer diameter than the first large-diameter eccentric part (41). Further, in

terms of degree of eccentricity with respect to the axis of the main spindle (**44**), the second large-diameter eccentric part (**42**) is larger than the first large-diameter eccentric part (**41**).

[0030] The compressor (**50**) is constituted by a rolling piston rotary compressor. The compressor (**50**) includes two cylinders (**51**, **52**) and two pistons (**57**). In the compressor (**50**), a rear head (**55**), the first cylinder (**51**), a middle plate (**56**), the second cylinder (**52**) and a front head (**54**) are stacked in bottom to top order.

[0031] In the inside of each of the first and second cylinders (**51**. **52**), a single cylindrical piston (**57**) is disposed. Though not shown, a tabular blade extends from the side surface of the piston (**57**) and is supported through a rolling bush to the associated cylinder (**51**, **52**). The piston (**57**) in the first cylinder (**51**) engages with the first lower eccentric part (**58**) of the shaft (**40**). On the other hand, the piston (**57**) in the second cylinder (**52**) engages with the second lower eccentric part (**59**) of the shaft (**40**). Each of the pistons (**57**, **57**) slides with its inner periphery on the outer periphery of the associated lower eccentric part (**58**, **59**) and slides with its outer periphery on the inner periphery of the associated cylinder (**51**, **52**). Thus, a compression chamber (**53**) is defined between the outer periphery of each of the pistons (**57**, **57**) and the inner periphery of the associated cylinder (**51**, **52**).

[0032] The first and second cylinders (**51**, **52**) are formed with single suction ports (**33**), respectively. Each suction port (**33**) radially passes through the associated cylinder (**51**, **52**) and opens at the distal end into the inner periphery of the cylinder (**51**, **52**). Further, each suction port (**33**) is extended through a pipe to the outside of the casing (**31**).

[0033] The front head (**54**) and the rear head (**55**) are formed with single discharge ports, respectively. The discharge port in the front head (**54**) brings the compression chamber (**53**) in the second cylinder (**52**) into communication with the inner space of the casing (**31**). The discharge port in the rear head (**55**) brings the compression chamber (**53**) in the first cylinder (**51**) into communication with the inner space of the casing (**31**). Further, each discharge port is provided at the distal end with a discharge valve formed of a lead valve and configured to be opened and closed by the discharge valve. In Figure **2**, the discharge

ports and discharge valves are not given. Gas refrigerant discharged from the compressor (**50**) into the inner space of the casing (**31**) is sent out through the discharge pipe (**36**) from the compression/expansion unit (**30**).

[0034] The expander (**60**) is constituted by a so-called rolling piston fluid machine. The expander (**60**) is provided with two cylinders (**71, 72**) and two pistons (**75, 85**) in two pairs. The expander (**60**) is also provided with a front head (**61**), a middle plate (**63**) and a rear head (**62**).

[0035] In the expander (**60**), the front head (**61**), the first cylinder (**71**), the middle plate (**63**), the second cylinder (**81**) and the rear head (**62**) are stacked in bottom to top order. In this state, the first cylinder (**71**) is closed at the lower end surface by the front head (**61**) and closed at the upper end surface by the middle plate (**63**). On the other hand, the second cylinder (**81**) is closed at the lower end surface by the middle plate (**63**) and closed at the upper end surface by the rear head (**62**). Further, the second cylinder (**81**) has a larger inner diameter than the first cylinder (**71**). The shaft (**40**) passes through the front head (**61**), the first cylinder (**71**), the middle plate (**63**), the second cylinder (**81**) and the rear head (**62**) that are in a stacked form.

[0036] As shown in Figures **3** and **4**, the first piston (**75**) and the second piston (**85**) are placed in the first cylinder (**71**) and the second cylinder (**81**), respectively. The first and second pistons (**75, 85**) are formed in an annular or cylindrical shape. The first piston (**75**) and the second piston (**85**) have equal outer diameters. The first piston (**75**) and the second piston (**85**) are passed through by the first large-diameter eccentric part (**41**) and the second large-diameter eccentric part (**42**), respectively. In the first cylinder (**71**), its inner periphery defines a first fluid chamber (**72**) together with the outer periphery of the first piston (**75**). In the second cylinder (**81**), its inner periphery defines a second fluid chamber (**82**) together with the outer periphery of the second piston (**85**).

[0037] The first and second pistons (**75, 85**) are integrally formed with single blades (**76, 86**), respectively. Each blade (**76, 86**) is formed into a plate extending radially from the

associated piston (75, 85) and extends outward from the outer periphery of the piston (75, 85).

[0038] The cylinders (71, 81) are provided with single pairs of bushes (77, 87), respectively. Each bush (77, 87) is a small piece formed so that its inside surface is plane and its outside surface is arcuate. Each pair of bushes (77, 87) are disposed with the associated blade (76, 86) sandwiched between them. Each bush (77, 87) slides with the inside surface on the associated blade (76, 86) and slides with the outside surface on the associated cylinder (71, 81). Each blade (76, 86) integral with the piston (75, 85) is supported through the associated bushes (77, 87) to the associated cylinder (71, 81) and free to angularly move about, extend into and retract from the cylinder (71, 81). The first cylinder (71) and the second cylinder (81) are arranged in postures in which the circumferential relative positions between their associated pairs of bushes (77, 87) are the same.

[0039] The first fluid chamber (72) in the first cylinder (71) is partitioned by the first blade (76) integral with the first piston (75); a region thereof to the left of the first blade (76) in Figure 4 provides a first high-pressure chamber (73) of relatively high pressure and a region thereof to the right of the first blade (76) provides a first low-pressure chamber (74) of relatively low pressure. The second fluid chamber (82) in the second cylinder (81) is partitioned by the second blade (86) integral with the second piston (85); a region thereof to the left of the second blade (86) in Figure 4 provides a second high-pressure chamber (83) of relatively high pressure and a region thereof to the right of the second blade (86) provides a second low-pressure chamber (84) of relatively low pressure.

[0040] The first cylinder (71) is formed with the inlet port (34). The inlet port (34) opens into the inner periphery of the first cylinder (71) slightly to the left of the bushes (77) in Figures 3 and 4. The inlet port (34) is communicable with the first high-pressure chamber (73). On the other hand, the second chamber (81) is formed with the outlet port (35). The outlet port (35) opens into the inner periphery of the second cylinder (81) slightly to the

right of the bushes (**87**) in Figures **3** and **4**. The outlet port (**35**) is communicable with the second low-pressure chamber (**84**).

[0041] The middle plate (**63**) is formed with a communicating channel (**64**). The communicating channel (**64**) passes through the middle plate (**63**) in the thickness direction. In the surface of the middle plate (**63**) facing the first cylinder (**71**), one end of the communicating channel (**64**) opens at a position to the right of the first blade (**76**). In the surface of the middle plate (**63**) facing the second cylinder (**81**), the other end of the communicating channel (**64**) opens at a position to the left of the second blade (**86**). Further, as shown in Figure **3**, the communicating channel (**64**) extends obliquely with respect to the thickness direction of the middle plate (**63**) and allows communication between the first low-pressure chamber (**74**) and the second high-pressure chamber (**83**).

[0042] In the expander (**60**) configured as described above, a first rotary mechanism (**70**) is constituted by the first cylinder (**71**) and the associated bushes (**77**), first piston (**75**) and first blade (**76**). Further, a second rotary mechanism (**80**) is constituted by the second cylinder (**81**) and the associated bushes (**87**), second piston (**85**) and second blade (**86**). Furthermore, in the expander (**60**), the displacement volume of the second rotary mechanism (**80**) (i.e., the maximum volume of the second fluid chamber (**82**)) is larger than that of the first rotary mechanism (**70**) (i.e., the maximum volume of the first fluid chamber (**72**)).

[0043] As can be seen from the above, in the expander (**60**), the timing when the first blade (**76**) retracts to the maximum to the outside of the first cylinder (**71**) is synchronized with the timing when the second blade (**86**) retracts to the maximum to the outside of the second cylinder (**81**). In other words, the stroke of the first rotary mechanism (**70**) during which the volume of the first low-pressure chamber (**74**) is decreasing is synchronized with the stroke of the second rotary mechanism (**80**) during which the volume of the second high-pressure chamber (**83**) is increasing (see Figure **4**). As described above, the first low-pressure chamber (**74**) of the first rotary mechanism (**70**) is communicated through the

communicating channel **(64)** with the second high-pressure chamber **(83)** of the second rotary mechanism **(80)**. The first low-pressure chamber **(74)**, the communicating channel **(64)** and the second high-pressure chamber **(83)** defines a single closed space and the closed space constitutes an expansion chamber **(66)**.

[0044] – Operational Behavior –

The above air conditioner **(10)** operates in a refrigeration cycle by circulating refrigerant through the refrigerant circuit **(15)**. In cooling operation, the refrigerant releases heat to the outdoor air sent to the gas cooler **(16)** and takes heat from the room air sent to the evaporator **(17)**. In this manner, the room air is cooled. On the other hand, in heating operation, the refrigerant releases heat to the room air sent to the gas cooler **(16)** and takes heat from the outdoor air sent to the evaporator **(17)**. In this manner, the room air is heated.

[0045] <Refrigeration Cycle in Refrigerant Circuit>

A description is given of a refrigeration cycle in the refrigerant circuit **(15)** with reference to the Mollier diagram (pressure-enthalpy diagram) of Figure **5**.

[0046] The refrigerant in a condition shown at point A in the figure is sucked into the compressor **(50)**. In the compressor **(50)**, the refrigerant in a condition shown at point A is compressed into a condition shown at point B. In the condition shown at point B, the pressure of refrigerant is higher than its critical pressure P_C. The refrigerant in the condition shown at point B is discharged from the compressor **(50)** and sent to the gas cooler **(16)**. In the gas cooler **(16)**, the refrigerant in the condition shown at point B releases heat to air and decreases its enthalpy while keeping the pressure constant, resulting in entering a condition shown at point C. During a transition period from the condition shown at point B from the condition shown at point C, the refrigerant temperature gradually decreases.

[0047] The refrigerant in the condition shown at point C flows out of the gas cooler **(16)** and is then introduced into the expander **(60)**. In the expander **(60)**, the refrigerant in the

condition shown at point C adiabatically expands and energy is recovered from the refrigerant. In the expander **(60)**, the refrigerant changes from the condition shown at point C to a condition shown at point D generally along an isentropic line. The refrigerant in the condition shown at point D flows out of the expander **(60)** and is then sent to the evaporator **(17)**. In the evaporator **(17)**, the refrigerant in the condition shown at point D takes heat from air and increases its enthalpy while keeping its pressure constant, resulting in entering the condition shown at point A. During a transition period from the condition shown at point D to the condition shown at point A, the refrigerant temperature is kept constant. The refrigerant in the condition shown at point A flows out of the evaporator **(17)**, is then sucked into the compressor **(50)** and compressed therein again.

[0048] <Behavior of Expander>

A description is given of the behavior of the expander **(60)** with reference to Figure **4**.

[0049] First, a description is given of the course of flow of supercritical high-pressure refrigerant into the first high-pressure chamber **(73)** of the first rotary mechanism **(70)**. When the shaft **(40)** rotates slightly from an angle of rotation of 0°, the contact point between the first piston **(75)** and the first cylinder **(71)** passes through the opening of the inlet port **(34)** so that high-pressure refrigerant begins to flow through the inlet port **(34)** into the first high-pressure chamber **(73)**. Then, as the angle of rotation of the shaft **(40)** gradually increases to 90°, 180° and 270°, high-pressure refrigerant flows more into the first high-pressure chamber **(73)**. The flow of the high-pressure refrigerant into the first high-pressure chamber **(73)** continues until the angle of rotation of the shaft **(40)** reaches 360°.

[0050] Next, a description is given of the course of refrigerant expansion in the expander **(60)**. When the shaft **(40)** rotates slightly from an angle of rotation of 0°, the first low-pressure chamber **(74)** is communicated through the communicating channel **(64)** with the second high-pressure chamber **(83)** so that the refrigerant begins to flow from the first low-

pressure chamber (**74**) into the second high-pressure chamber (**83**). Then, as the angle of rotation of the shaft (**40**) gradually increases to 90°, 180° and 270°, the first low-pressure chamber (**74**) gradually decreases its volume and, concurrently, the second high-pressure chamber (**83**) gradually increases its volume, resulting in gradually increasing volume of the expansion chamber (**66**). The increase in the volume of the expansion chamber (**66**) continues until the angle of rotation of the shaft (**40**) reaches 360°. The refrigerant in the expansion chamber (**66**) expands while decreasing its pressure during the increase in the volume of the expansion chamber (**66**). Further, the internal pressure difference between the first high-pressure chamber (**73**) and the first low-pressure chamber (**74**) and the internal pressure difference between the second high-pressure chamber (**83**) and the second low-pressure chamber (**84**) produce torque. The torque drives the shaft (**40**) into rotation. Thus, the refrigerant in the first low-pressure chamber (**74**) flows through the communicating channel (**64**) and then into the second high-pressure chamber (**83**) while expanding.

[0051] Finally, a description is given of the course of flow of refrigerant from the second low-pressure chamber (**84**) of the second rotary mechanism (**80**). The second low-pressure chamber (**84**) starts to be communicated with the outlet port (**35**) from a point of time when the shaft (**40**) is at an angle of rotation of 0°. In other words, from that point of time, the refrigerant starts to flow out of the second low-pressure chamber (**84**) to the outlet port (**35**). Then, during the period when the angle of rotation of the shaft (**40**) gradually increases to 90°, 180° and 270° and until it reaches 360°, low-pressure refrigerant obtained by expansion flows out of the second low-pressure chamber (**84**).

[0052] – Specification of Expander –

As described above, in the expander (**60**), the second rotary mechanism (**80**) has a larger displacement volume than the first rotary mechanism (**70**). The displacement volume of the first rotary mechanism (**70**) is the maximum volume of the first fluid chamber (**72**),

i.e., the volume of the first fluid chamber (**72**) just after the closing off of fluid communication from the inlet port (**34**). The displacement volume of the second rotary mechanism (**80**) is the maximum volume of the second fluid chamber (**82**), i.e., the volume of the second fluid chamber (**82**) just before the provision of fluid communication with the outlet port (**35**).

[0053] A description is given here of how the displacement volume of each rotary mechanism (**70, 80**) in the expander (**60**) is set.

[0054] In the air conditioner (**10**), the refrigerant in the gas cooler (**16**) and the evaporator (**17**) exchanges heat with room air or outdoor air. For this purpose, consideration is made of room air and outdoor air conditions to set the low-side pressure of the refrigeration cycle (reference low-pressure P_L) and the refrigerant temperature at the exit of the gas cooler (**16**) (reference refrigerant temperature T_2) under reference operating conditions serving as a design standard. Further, if the air conditioning capacity desired for the air conditioner (**10**) is assumed, the necessary amount of refrigerant (the flow rate of refrigerant) circulated through the refrigerant circuit (**15**) to obtain the assumed air conditioning capacity is determined. The suction volume v_1 of the compressor is set accordingly. Furthermore, since the expander (**60**) is connected to the compressor (**50**) by a single shaft (**40**), the expander (**60**) and the compressor (**50**) have the same number of rotations. In other words, the rotational speed ratio r of the compressor (**50**) to the expander (**60**) is "1".

[0055] As described above, in a refrigeration cycle having a high-side refrigerant pressure equal to or above the critical pressure of the refrigerant (in a so-called supercritical cycle), the high-side pressure of the refrigeration cycle cannot be determined only in consideration of room air and outdoor air conditions. On the other hand, as shown in Figure **6**, if in a supercritical cycle the low-side pressure of the refrigeration cycle and the refrigerant temperature at the exit of the gas cooler (**16**) are fixed, the COP (coefficient of performance) of the refrigeration cycle varies according to the high-side pressure of the

refrigeration cycle. When the high-side pressure reaches a particular value, the COP of the refrigeration cycle reaches its maximum value.

[0056] The ordinate of the graph shown in Figure **6** indicates the COP in the case where the object is heated by the gas cooler **(16)** and the COP is expressed by $\Delta h_{BC}/(\Delta h_{BA}-\Delta h_{CD})$. As shown in Figure **5**, Δh_{BC} denotes the amount of heat given from refrigerant to the object in the gas cooler **(16)**, Δh_{BA} denotes the energy required to compress the refrigerant in the compressor and Δh_{CD} denotes the energy recovered from the refrigerant in the expander, all of them being expressed per kg of refrigerant.

[0057] In the expander **(60)** of the present embodiment, attention is focused on a characteristic of the supercritical cycle that "if the low-side pressure of a refrigeration cycle and the refrigerant temperature at the exit of the gas cooler **(16)** are fixed, the high-side pressure at which the COP reaches its maximum value is uniquely determined", and this characteristic is used to set the displacement volume of each rotary mechanism **(70, 80)**.

[0058] A detailed description is given of the above point. Assuming that the compressor **(50)** sucks saturated gas refrigerant, the density of refrigerant sucked in the compressor **(50)** is the density ρ_1 of saturated gas refrigerant at the reference low pressure. The amount of refrigerant discharged per each rotation of the compressor **(50)** is expressed by the value ($\rho_1 v_1$) obtained by multiplying the density ρ_1 of saturated gas refrigerant at the reference low pressure by the suction volume v_1 of the compressor. Assume that ρ_2 is the density of refrigerant at the reference high pressure and the reference refrigerant temperature, i.e., the density of refrigerant introduced into the expander **(60)**. Ideally speaking, the amount of refrigerant introduced into the expander **(60)** should be equal to the amount of refrigerant discharged from the compressor **(50)**. Therefore, the displacement volume of the first rotary mechanism **(70)**, or the maximum volume v_2 of the first fluid chamber **(72)**, is set at $v_2=\rho_1 v_1 r/\rho_2$.

[0059] Assuming that the refrigerant adiabatically expands in the expander **(60)**, the

refrigerant density ρ_3 at the exit of the expander **(60)** is determined. The amount of refrigerant flowing out of the expander **(60)** is always equal to the mount of refrigerant introduced into the expander **(60)**. Therefore, the displacement volume of the second rotary mechanism **(80)**, or the maximum volume v_3 of the second fluid chamber **(82)**, is set at $v_3=\rho_2 v_2/\rho_3$.

[0060] A concrete example of the above detailed description is described below. A description is given here of the case where, under reference operating conditions serving as a design standard, the refrigerant evaporation temperature is set at 0°C and the refrigerant temperature at the exit of the gas cooler **(16)** is set at 35°C. In this case, the reference refrigerant temperature is set at 35°C. The reference low pressure is set at a pressure of 3.5MPa at which the evaporation temperature of carbon dioxide (CO_2) serving as the refrigerant reaches 0°C. The density ρ_1 of saturated gas refrigerant at a reference low pressure of 3.5MPa is 97.32 kg/m^3. Further, under the operating conditions of a reference low pressure of 3.5MPa and a reference refrigerant temperature of 35°C, the high-side pressure at which the COP of the refrigeration cycle reaches its maximum, i.e., the reference high pressure, is 9MPa (see Figure **6**). The refrigerant density ρ_2 at a reference high pressure of 9MPa and a reference refrigerant temperature of 35°C is 662.5 kg/m^3. Further, the density ρ_3 of refrigerant adiabatically expanded from a condition of a reference high pressure of 9MPa and a reference refrigerant temperature of 35°C into a condition of a refrigerant low pressure of 3.5MPa is 220 kg/m^3. Therefore, the displacement volume v_2 of the first rotary mechanism **(70)** is set at $v_2=(97.32/662.5)v_1=0.15v_1$ and the displacement volume v_3 of the second rotary mechanism **(80)** is set at $v_3=(662.5/220)v_2=0.44v_1$.

[0061] – Effects of Embodiment 1 –

In the present embodiment, attention is focused on a characteristic of the supercritical cycle that under conditions in which the low-side pressure of a refrigeration cycle (the reference low pressure) and the refrigerant temperature at the exit of the gas

cooler (16) (the reference refrigerant temperature) are fixed, the high-side pressure of the refrigeration cycle (the reference high pressure) at which the COP reaches its maximum value is uniquely determined, and the expander (60) specified using the above characteristic is provided in the air conditioner (10) serving as a refrigeration system. Therefore, according to the present embodiment, the expander (60) having an optimal specification for operating conditions of the air conditioner (10) can be used and the energy recovered from the refrigerant by the expander (60) can be increased to enhance the COP of the air conditioner (10).

[0062] <<Embodiment 2 of the Invention>>

A description is given of Embodiment 2 of the present invention. An air conditioner (10) of the present embodiment is described here in terms of differences from that of Embodiment 1.

[0063] As shown in Figure 7, the refrigerant circuit (15) of the air conditioner (10) is provided with a receiver (18). The receiver (18) is formed in the shape of a cylindrical closed container and disposed between the exit side of the evaporator (17) and the suction side of the compressor (50). Within the receiver (18), part of refrigerant charged in the refrigerant circuit (15) is stored in the form of liquid refrigerant. When the amount of liquid refrigerant in the receiver (18) increases or decreases, the amount of refrigerant circulating through the refrigerant circuit (15) varies accordingly.

[0064] In the refrigerant circuit (15) of the present embodiment, the refrigerant having flowed out of the evaporator (17) is introduced into the receiver (18) and the refrigerant in the receiver (18) is sucked into the compressor (50). Since liquid refrigerant exists within the receiver (18), the refrigerant in the receiver (18) to be sucked into the compressor (50) is in a saturated state. Therefore, even under an operating condition in which the refrigerant is in a superheated state at the exit of the evaporator (17), the refrigerant to be sucked into the compressor (50) is kept saturated.

[0065] As already described in the description of Embodiment 1, the displacement volume of each rotary mechanism (**70**, **80**) in the expander (**60**) is set on the assumption that the refrigerant to be sucked into the compressor (**50**) is saturated gas refrigerant. Therefore, in the case where the receiver (**18**) is disposed to keep the refrigerant to be sucked into the compressor (**50**) in a saturated state regardless of operating conditions, the conditions of the refrigeration cycle in the refrigerant circuit (**15**) can be approximated to operating conditions expected in designing the refrigeration system, which enables stabilization of the refrigeration cycle in the refrigerant circuit (**15**).

[0066] <<Embodiment 3 of the Invention>>

A description is given of Embodiment 3 of the present invention. An air conditioner (**10**) of the present embodiment is described here in terms of differences from that of Embodiment 2.

[0067] As shown in Figure **8**, the refrigerant circuit (**15**) of the air conditioner (**10**) is provided with an internal heat exchanger (**20**). The internal heat exchanger (**20**) includes a first channel (**21**) and a second channel (**22**). The first channel (**21**) is connected on the entrance side to the gas cooler (**16**) and connected on the exit side to the inlet port (**34**) of the expander (**60**). The second channel (**22**) is connected on the entrance side to the evaporator (**17**) and connected on the exit side through the receiver (**18**) to the suction port (**32**) of the compressor (**50**).

[0068] In the internal heat exchanger (**20**), the refrigerant flowing from the gas cooler (**16**) towards the expander (**60**) is cooled by heat exchange with the refrigerant flowing from the evaporator (**17**) towards the receiver (**18**). The refrigerant flowing towards the expander (**60**) decreases its enthalpy by being cooled in the internal heat exchanger (**20**). The refrigerant sent from the expander (**60**) to the evaporator (**17**) also decreases its enthalpy accordingly. Therefore, the refrigerant enthalpy difference between the entrance and exit of the evaporator (**17**) can be increased, thereby increasing the amount of heat

taken from air by the refrigerant in the evaporator (**17**). Hence, according to the present embodiment, the provision of the internal heat exchanger (**20**) provides enhanced cooling capacity of the air conditioner (**10**).

[0069] – Modification of Embodiment 3 –

In the refrigerant circuit (**15**) of the present embodiment, as shown in Figure **9**, the second channel (**22**) of the internal heat exchanger (**20**) may be connected between the receiver (**18**) and the compressor (**50**). Specifically, in the present modification, the second channel (**22**) of the internal heat exchanger (**20**) is connected on the entrance side through the receiver (**18**) to the evaporator (**17**) and connected on the exit side to the suction port (**32**) of the compressor. Also in the internal heat exchanger (**20**) of the present modification, the refrigerant flowing from the gas cooler (**16**) towards the expander (**60**) is cooled by heat exchange with the refrigerant flowing from the evaporator (**17**) towards the compressor (**50**), which increases the refrigerant enthalpy difference between the entrance and exit of the evaporator (**17**).

[0070] <<Embodiment 4 of the Invention>>

A description is given of Embodiment 4 of the present invention. The present embodiment differs in the structure of the expander (**60**) from Embodiment 1.

[0071] As shown in Figure **10**, the expander (**60**) in the present embodiment is constituted by a scroll type fluid machine. The expander (**60**) includes a movable scroll (**91**) and a fixed scroll (**93**). The movable scroll (**91**) includes a movable wrap (**92**). The movable wrap (**92**) is formed in the shape of a spiral wall the top of which describes an involute curve. The movable scroll (**91**) is engaged with the shaft (**40**) and configured to only move bodily around the shaft (**40**) while being held against rotation on its axis. The fixed scroll (**93**) includes a fixed wrap (**94**). The movable wrap (**92**) is formed in the shape of a spiral wall corresponding to the movable wrap (**92**) and both the side surfaces form envelopes for the fixed wrap (**94**) moving bodily around the shaft (**40**). In the fixed scroll

(93), the inlet port (34) opens into its central portion and the outlet port (35) opens into its peripheral portion.

[0072] In the expander (60), the movable wrap (92) of the movable scroll (91) is intermated with the fixed wrap (94) of the fixed scroll (93). Between the movable wrap (92) and the fixed wrap (94), a first chamber (95) and a second chamber (96), both servings as fluid chambers, are formed as a pair. As shown sequentially in Figures **10(A)** to **10(D)**, as the movable scroll (91) moves, the first chamber (95) and the second chamber (96) change their volumes.

[0073] Figure **10(A)** shows a state of the expander just after the closing off of fluid communication of the first chamber (95) and the second chamber (96) from the inlet port (34). In this state, the first chamber (95) and the second chamber (96) have minimum volumes. In the expander (60) of the present embodiment, the sum of the volumes of the first chamber (95) and the second chamber (96) in this state is v_2. On the other hand, Figure **10(D)** shows a state of the expander just before the provision of fluid communication of the first chamber (95) and the second chamber (96) with the outlet port (35). In this state, the first chamber (95) and the second chamber (96) have maximum volumes. In the expander (60) of the present embodiment, the sum of the volumes of the first chamber (95) and the second chamber (96) in this state is v_3.

[0074] The embodiments described so far are merely preferred illustrative embodiments of the invention in nature and are not intended to limit the scope, applications and use of the invention.

Industrial Applicability

[0075] As seen from the above, the present invention is useful as a refrigeration system that includes an expander and operates in a refrigeration cycle having a high-side refrigerant pressure set equal to or above the critical pressure of the refrigerant.